

Optimization of design radiation for concentrating solar thermal power plants without storage

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Abstract

Prior to the detailed design of a concentrating solar power (CSP) plant, it is necessary to develop a conceptual design of the plant. Conceptual design activities help in screening of various design objectives prior to the detailed design of the overall power plant. Conceptual design of a CSP plant includes type and size of solar field, power generating cycle and the working fluid, sizing of the power block, etc. One of the most important parameters for conceptual design of a CSP plant is to determine the design radiation of the plant. Design radiation for a CSP plant is the direct normal irradiance (DNI) at which the plant produces the rated power output. Due to daily and seasonal variation of radiation, determining appropriate design radiation is extremely important; low design radiation results in excessive unutilized energy and high design radiation results in low capacity factor of the plant. In this paper, a methodology is proposed to determine the thermodynamically and the cost optimum design radiation for CSP plants without hybridization and storage. The proposed methodology accounts for the characteristics of the collector field as well as turbine characteristics. There is no such analytical methodology reported in the literature. Applicability of the proposed methodology is demonstrated through illustrative case studies incorporating parabolic trough collectors (PTC) as well as linear Fresnel reflectors (LFR). Design radiations for ten different places in India are also reported. However, the proposed methodology is not restricted to India only.

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1. Introduction

Technical feasibility of producing high-temperature steam using concentrated solar power (CSP) and to produce power through conventional Rankine cycle has already been successfully demonstrated. Today, there are solar thermal power plants based on CSP technology with electrical power generation capacity of few kW to 50 MW and more in various sun rich regions around the world (Fernández-García et al., 2010). There are mainly four

commercially available CSP technologies. Two of them, parabolic trough collector (PTC) and linear Fresnel reflector (LFR), concentrate direct radiation onto a line, whereas the other two, paraboloid dish and solar power tower (SPT) technologies, concentrate direct radiation onto a point. Comparative analysis between PTC and LFR based CSP plants have been presented by Morin et al. (2012) and Giottri et al. (2012a). Franchini et al. (2013) have presented the comparative analysis between PTC and SPT based CSP plants. Detailed studies on economic aspects of CSP plants have been reported by Krishnamurthy et al. (2012) and Kost et al. (2013). Among the CSP technologies, plants with PTC, using synthetic or organic oil as heat transfer fluid

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Nomenclature

a	Willans' line equation parameter (W)
A_p	aperture area of the collector (m^2)
b	Willans' line equation parameter (J/kg)
C_p	specific heat (J/kg)
E	aperture specific energy output (Wh/m^2)
h	enthalpy (J/kg)
I	aperture effective direct normal irradiance (W/m^2)
L	loss (W/m^2)
m	mass flow rate (kg/s)
P	power (W)
Q	heat (W)
T	temperature ($^{\circ}\text{C}$)
U_L	heat loss coefficient based on aperture area ($\text{W}/\text{m}^2 \text{ K}$)
x	dryness fraction
y	fraction of internal losses of turbine

Greek symbols

Δ	difference
η	efficiency
θ	incidence angle ($^{\circ}$)
τ	constant

Abbreviations

CSP	concentrating solar power
DNI	direct normal irradiance
DSG	direct steam generation
HTF	heat transfer fluid
LFR	linear Fresnel reflector
PTC	parabolic trough collector
SAM	system advisor model
SPT	solar power tower

Subscripts

a	ambient
C	threshold
CL	collector
D	design
hx	heat exchanger
is	isentropic
m	mean
max	maximum
min	minimum
o	optical
w	water

(HTF), are found to be more attractive commercially (Purohit et al., 2013). In such plants, the maximum temperature is limited to about 400°C with a resulting steam temperature at the turbine inlet about 350 – 370°C .

Use of molten salt as a HTF can raise the steam temperature up to 540°C , allowing steam turbines to operate at greater efficiency (Montes et al., 2010; Giostri et al., 2012b; Yang and Garimella, 2013; Zaversky et al., 2013). It may be noted that the direct steam generation (DSG) in PTC field is also an economically viable option (Odeh et al., 1998; Zarza et al., 2002; Eck and Steinmann, 2005; Eck and Hirsch, 2007; Hirsch et al., 2014). Several investigators have proposed DSG in LFR field as cheaper alternative due to the use of flat mirrors and simpler structures, though with a lower optical efficiency (Mills and Morrison, 2006; Xie et al., 2012; Sahoo et al., 2012, 2013; Zhu, 2013). LFR based CSP plants have been tested and installed on a utility scale around the world (Facão and Oliveira, 2011; Abbas et al., 2012; Zhu et al., 2014). Under the initiative of Indian Institute of Technology Bombay (IIT Bombay), a one megawatt CSP plant was proposed in the year 2009 and currently being commissioned in India that integrates a LFR field for DSG and PTC field for HTF (Desai et al., 2013). HTF is used for both steam generation and superheating the combine steam (Desai et al., 2013). A similar configuration has been proposed by Manzolini et al. (2011) where the PTC field is divided into two sections: one generates saturated steam

like in DSG process and the second heats up a conventional HTF which is used for superheating and reheating the steam.

SPT is the second most installed CSP plant technology after PTC, and gradually gaining acceptance (Avila-Marin et al., 2013; Zhang et al., 2013). Some commercial SPT plants now in operation use DSG (Müller-Steinhagen and Trieb, 2004) and other plants use molten salts as HTF as well as storage medium (Amadei et al., 2013; Cáceres et al., 2013). A detailed review on design of central receiver and SPT based CSP plants have been reported by Behar et al. (2013) as well as Ho and Iverson (2014), respectively. Non-traditional heliostat field layouts (e.g., on hillsides) has also been reported in the literatures (Noone et al., 2011; Slocum et al., 2011). A paraboloid dish can be used for operating a Stirling engine located at its focus, DSG or linked with other dishes to heat a transfer fluid which is then used to drive a conventional turbine (Wu et al., 2010; Siva Reddy et al., 2013; Reddy and Veershetty, 2013). It is interesting to note that paraboloid dish systems give the highest efficiency among CSP technologies (Sharma, 2011).

Conventional Rankine cycle which uses water/steam as working fluid is most widely used in the CSP plants. With growing interest for highly efficient and modular CSP plants (Quoilin et al., 2011; Casati et al., 2013), organic Rankine cycles (i.e., a Rankine cycle that uses an organic fluid as the working medium) are very promising due to a

number of advantages over the steam cycle (Desai and Bandyopadhyay, 2009; Tchanche et al., 2011). Organic Rankine cycle can also be used as a bottoming cycle in CSP plants (Al-Sulaiman, 2014). Mittelman and Epstein (2010) have presented a configuration with Kalina cycle as a bottoming cycle in a CSP plant. Jamel et al. (2013) have reported the advances in integrating solar thermal energy with conventional and non-conventional power plants.

Intensive research has been conducted on modeling and simulation of CSP plants. Probabilistic modeling of CSP plants has been presented by Ho et al. (2011) and Zaversky et al. (2012). Dynamic simulation models for CSP plants, with thermal energy storage, have also been proposed (García et al., 2011; Powell and Edgar, 2012). Exergetic analysis of PTC based CSP plant has been presented by Siva Reddy et al. (2012). Several parabolic trough system models have been developed, including physical trough model of System Advisor Model (SAM), to predict the hourly performance of solar field (Wagner et al., 2010). PATTO code can predict the overall performance of PTC based CSP plants (Manzolini et al., 2011). SimulCET simulates energy absorption in the PTC plant, based on both empirical and analytical expressions (García-Barberena et al., 2012). Solar thermal simulator (IIT Bombay, 2012) can be used for preliminary sizing, heat balance design, off-design simulations and performance evaluation of a complete plant or a small subset of complete plant (Desai et al., 2013). Detailed comparisons of solar thermal simulator (IIT Bombay, 2012) with different widely used software, like SAM, TRNSYS and Thermoflex, for solar thermal application have been reported by Desai and Bandyopadhyay (2012).

Prior to the detailed design of a CSP plant, it is necessary to develop a conceptual design of the plant. It is important to screen numerous design alternatives prior to the detailed design. Conceptual design of a CSP plant includes type and size of solar field, power generating cycle and the working fluid, sizing of the power block, etc. One of the most important parameters during conceptual design of a CSP plant is the design radiation. Design radiation for a CSP plant is the direct normal irradiance (DNI) at which

the plant produces the rated power output. Fig. 1 shows the daily variation of DNI with two different design conditions. Due to daily and seasonal variation of radiation, determining appropriate design radiation is extremely important; low design radiation results in excessive unutilized energy and high design radiation results in low capacity factor of the plant, resulting poor utilization of the invested capital. Therefore, there exists an optimal design DNI for CSP plant without hybridization and thermal storage. However, it may be noted that once place is fixed, the design radiation depends on collector field and turbine characteristic parameters.

Quaschnig et al. (2002) have proposed a method for estimating the optimized solar field size as a function of solar irradiance and economic aspects through multiple simulations. An approximate method to find the cost optimum design radiation for a parabolic trough CSP plant, without hybridization and thermal storage, based on frequency distribution of DNI, corrected by cosine of the incidence angle, has also been proposed by Quaschnig et al. (2002). However, the effects of solar collector parameters (except cosine of the incidence angle) as well as turbine characteristic parameters are not considered. Montes et al. (2009) have presented an economic optimization of design radiation for parabolic trough solar thermal power plant without hybridization and thermal storage through multiple simulations. Sundaray and Kandpal (in press) have studied the effect of design DNI on capacity factor and unutilized energy using SAM. In cases where detailed models are used for simulation, the total number of equations can be significantly high, depending on the number of components used. Apart from these, due to the complexity of plant description and number of recursive dependencies between various equipments, the computational effort to calculate the model response at every operating condition for a typical operation year (with 8760 calculation) would be very high. In this paper, a simple methodology is proposed to find the thermodynamically optimum design radiation, for which the annual power generation per unit aperture area of collector is the maximum, using collector and turbine characteristic parameters for CSP plant without hybridization and thermal energy storage. The proposed methodology does not require any simulation. A methodology to determine optimum design radiation that minimizes cost of electricity generation is also proposed. There is no such analytical methodology reported in the literature. The validity of method is demonstrated through illustrative case studies of PTC based and LFR based CSP plants. The proposed methodology can be applied to CSP plants with any technology.

2. Effect of design DNI on annual power output

It is possible to select an appropriate value of DNI at the conceptual design of CSP plants so that a proper tradeoff between the capacity factor and unutilized energy is achieved. In case of CSP plants without hybridization

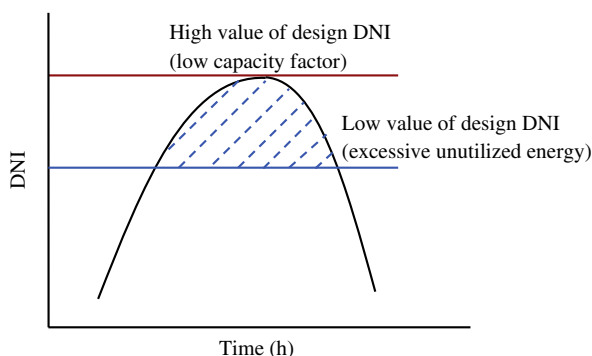


Fig. 1. Daily variation of DNI and two different design conditions.

Table 1
Data used for the simulation.

Input parameter	Value/type
Place	Jodhpur (India)
Collector field	Parabolic Trough Collector (PTC)
Collector aperture area	10,000 m ² (fixed value)
Collector field efficiency model parameters	Collector field efficiency: $\eta = \eta_o - U_L \cdot \left(\frac{T_m - T_a}{\text{DNI} \cdot \cos(\theta)} \right)$ where $\eta_o = 0.7$; $U_L = 0.1$ (W/m ² K)
Collector tracking mode	Focal axis N–S horizontal and E–W tracking
Collector outlet temperature	390 °C (controlled)
Ambient temperature	30 °C (design value)
Willans' line equation	Turbine power output: $P = a + b \cdot m$ where $a = -0.2 \cdot P_D$ (Mavromatis and Kokossis, 1998) $b = 786$ kJ/kg
Turbine inlet pressure	40 bar
Turbine inlet temperature	350 °C (design value)
Turn down ratio of the turbine (P_{min}/P_{max})	0.2
Temperature driving force (ΔT_{min})	For heat exchanger = 10 °C (design value) For condenser = 5 °C
Isentropic efficiency of the pump	60%
Condensing pressure	0.1 bar

and thermal storage, the choice of design DNI depends on location and technology selected. To illustrate the existence of optimal design DNI, simulations of a PTC based CSP plant, without storage and auxiliary boiler, are performed. Data used for simulation are given in Table 1. DNI data for the simulation are taken from Ramaswamy et al. (2013). Simulations are performed in the solar thermal simulator (IIT Bombay, 2012) and the schematic of such a plant is shown in Fig. 2.

Annual turbine and net power outputs, at different design DNI, obtained from the solar thermal simulator (IIT Bombay, 2012), are provided in Table 2. The maximum annual turbine output of 2222.3 MW h and the maximum net annual output of 2183 MW h are obtained with a design DNI of 680 W/m². To meet the required

power output from the plant at any design DNI, the turbine output can be fixed and the collector aperture area can be varied. It may be noted that, the optimum design DNI remains identical to the previous result. This is equivalent to the methods used by Montes et al. (2009) as well as Sundaray and Kandpal (in press). It may also be noted from Table 2 that nature of the net power output curve is not very sharp near the maximum. Design DNI within the range 630–730 W/m², the net annual output of the plant remains within 1% of the maximum. Similar observations are reported for many practical optimizations problems, e.g., see Shenoy et al. (1998) for optimization of heat exchanger networks.

3. Approximate method to determine optimal design DNI

Due to daily and seasonal variation of radiation, the CSP plant without hybridization and storage operates at part loads most of the time. Willans' line provides a linear relation between the steam flow rate and turbine power output. This is very useful in predicting the performance of steam turbine at part loads. The power output of turbine can be calculated based on Willans' line equation as:

$$P = a + b \cdot m \quad (1)$$

and the design power output (P_D) may be written as:

$$P_D = a + b \cdot m_D \quad (2)$$

In Eqs. (1) and (2), P and P_D are power output at steam flow rate m and m_D , respectively. It may be noted that a and b are Willans' line parameters. Assuming that internal losses amount to y times of the rated output of the turbine (Mavromatis and Kokossis, 1998):

$$a = -y \cdot P_D \quad (3)$$

Substituting the value of a in Eqs. (1) and (2):

$$P = -y \cdot P_D + b \cdot m \quad (4)$$

$$P_D = -y \cdot P_D + b \cdot m_D \quad (5)$$

From Eqs. (4) and (5):

$$P_D - P = b \cdot (m_D - m) \quad (6)$$

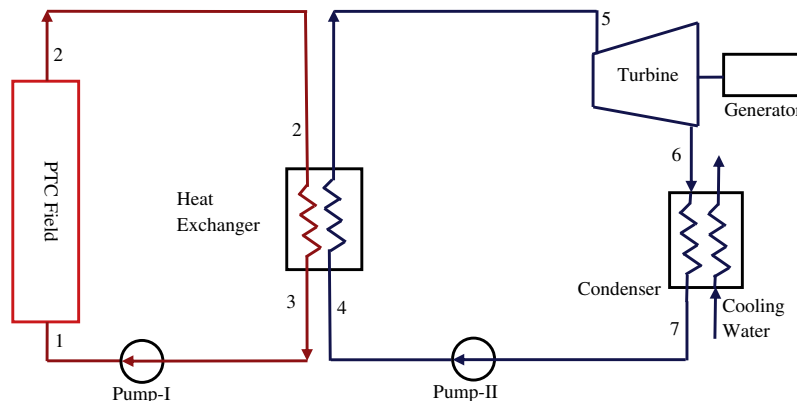


Fig. 2. Simplified process flow diagram of a PTC based CSP power plant without hybridization and thermal storage.

Table 2

Simulation results for a PTC based solar thermal power plant without hybridization and thermal storage.

Design DNI (W/m ²)	Design mass flow rate of oil (kg/s)	Design mass flow rate of steam (kg/s)	Design turbine output (MW)	Annual turbine output (MW h/y)	Net annual output (MW h/y)
550	7.89	1.240	0.812	2098.5	2065.6
600	8.66	1.361	0.891	2175.0	2139.2
620	8.97	1.409	0.923	2195.8	2158.8
630	9.12	1.433	0.939	2203.9	2166.6
660	9.58	1.506	0.986	2220.2	2181.7
670	9.74	1.530	1.008	2221.3	2182.3
680	9.89	1.554	1.026	2222.3	2183.0
690	10.05	1.578	1.034	2219.9	2180.3
700	10.20	1.603	1.050	2216.7	2176.8
730	10.66	1.675	1.097	2206.1	2165.8
740	10.82	1.699	1.113	2200.8	2160.4
800	11.74	1.845	1.208	2143.8	2103.2

The DNI corrected by cosine of the incidence angle (product of DNI and $\cos(\theta)$) is known as aperture effective DNI (Feldhoff et al., 2012). If it is denoted as I then the steam mass flow rate can be represented as a function of aperture effective DNI:

$$m_D = (\alpha I_D - \beta) \cdot A_p \text{ and } m = (\alpha I - \beta) \cdot A_p \quad (7)$$

where α and β are mainly dependent on collector and power generating cycle parameters. Values of these parameters are derived in subsequent sections. From Eqs. (6) and (7):

$$\frac{P_D - P}{A_p} = \tau \cdot (I_D - I) \quad (8)$$

where

$$\tau = b \cdot \alpha = \text{Constant} \quad (9)$$

The design power output can be calculated from the relation given below:

$$P_D = m_D \cdot \Delta h_{is} \cdot \eta_{is,D} \quad (10)$$

where $\eta_{is,D}$ and Δh_{is} are the isentropic efficiency of the turbine at design condition and isentropic enthalpy change in the turbine, respectively. Combining Eqs. (5) and (10):

$$b = (1 + y) \cdot \frac{P_D}{m_D} = (1 + y) \cdot \Delta h_{is} \cdot \eta_{is,D} \quad (11)$$

The value of design power output can also be calculated from Eq. (5):

$$P_D = \frac{b \cdot m_D}{(1 + y)} \quad (12)$$

From Eq. (9):

$$P_D = \frac{\tau \cdot m_D}{\alpha \cdot (1 + y)} \quad (13)$$

Substituting the value of m_D from Eq. (7), we get:

$$\frac{P_D}{A_p} = \frac{\tau \cdot (\alpha \cdot I_D - \beta)}{\alpha \cdot (1 + y)} \quad (14)$$

$$\frac{P_D}{A_p} = (\alpha' \cdot I_D - \beta') \quad (15)$$

where

$$\alpha' = \frac{\tau}{(1 + y)} \quad \text{and} \quad \beta' = \frac{\tau \cdot \beta}{(1 + y) \cdot \alpha} \quad (16)$$

Only the insolation that is directly normal to collector surface can be focused and thus, be available to heat the absorber tubes. The angle of incidence (θ) is the angle between the plane normal to a surface and the beam radiation on that surface. Along with the DNI, the angle of incidence also has diurnal and seasonal variations. These variations heavily influence the performance of collectors and hence, the performance of the overall CSP plant. The effective insolation available for concentrating collector is the product of DNI and $\cos(\theta)$, also known as aperture effective DNI. To capture the effect of the variation of aperture effective DNI on the plant performance, duration curve of aperture effective DNI is generated. Duration curve for the aperture effective DNI is generated in such a way that the highest value is plotted in left and the smallest one in right. A typical duration curve for the aperture effective DNI is shown in Fig. 3. On time axis, the time duration for which each certain value of aperture effective DNI continues during entire year is plotted. It enables us to determine the number of hours per year when aperture effective DNI is greater or lesser than any given value.

For a given I_D , energy loss due to part-load operation of turbine will be τ times the shaded area on lower side of I_D . Let, the value of threshold I , corresponding to minimum power output (P_{min}), is I_C . Loss due to part load operation of turbine (L) per unit aperture area of collector is:

$$L = \int_{f(I_D)}^{f(I_C)} \frac{(P_D - P)}{A_p} \cdot dt \quad (17)$$

From Eq. (8):

$$L = \int_{f(I_D)}^{f(I_C)} \tau \cdot (I_D - I) \cdot dt \quad (18)$$

Assuming the value of τ and $f(I_C)$ to be constant:

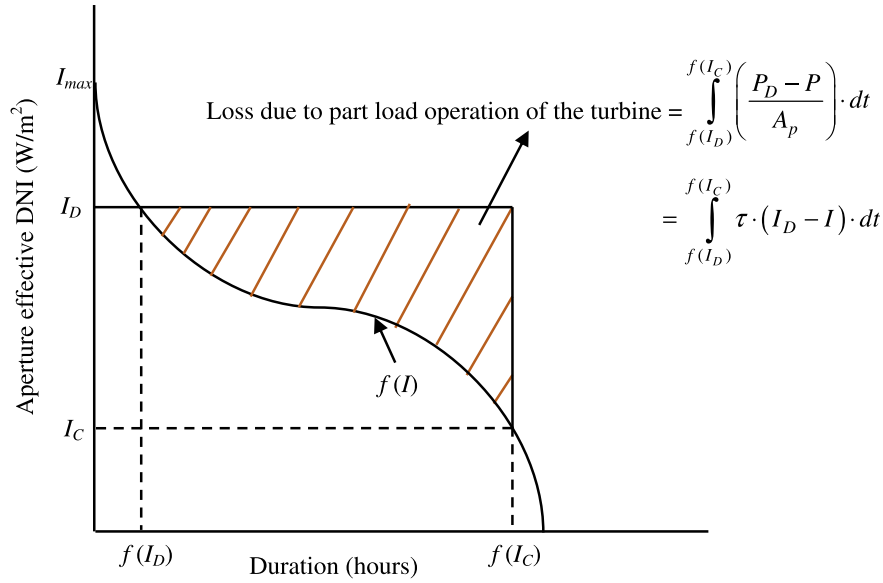


Fig. 3. A typical duration curve for aperture effective DNI and part load losses.

$$L = \tau \cdot \left(f(I_C) \cdot (I_D - I_C) - \int_{I_C}^{I_D} f(I) \cdot dI \right) \quad (19)$$

Annual turbine output (E) per unit aperture area of collector can be given as follows:

$$E = \left(\frac{P_D}{A_p} \cdot f(I_C) \right) - \tau \cdot \left(f(I_C) \cdot (I_D - I_C) - \int_{I_C}^{I_D} f(I) \cdot dI \right) \quad (20)$$

It may be noted that the turbine can operate at little higher flow rates than design. However, this is allowed for only some small fraction of time and therefore, this effect is neglected in the present study. From Eq. (15):

$$E = (\alpha' \cdot I_D - \beta') \cdot f(I_C) - \tau \cdot \left(f(I_C) \cdot (I_D - I_C) - \int_{I_C}^{I_D} f(I) \cdot dI \right) \quad (21)$$

Taking the derivative of annual turbine output with respect to I_D :

$$\frac{dE}{dI_D} = \alpha' \cdot f(I_C) - \tau \cdot f(I_C) + \tau \cdot f(I_D) \quad (22)$$

Taking the double derivative:

$$\frac{d^2E}{dI_D^2} = \tau \cdot f'(I_D) \quad (23)$$

As the value of $f'(I_D)$ is always lesser than zero and τ is always greater than zero:

$$\frac{d^2E}{dI_D^2} < 0 \quad (24)$$

Therefore, the condition for the maximum annual turbine output is:

$$\frac{dE}{dI_D} = \alpha' \cdot f(I_C) - \tau \cdot f(I_C) + \tau \cdot f(I_{D,optimum}) = 0 \quad (25)$$

$$f(I_{D,optimum}) = \frac{\tau - \alpha'}{\tau} \cdot f(I_C) \quad (26)$$

Substituting the value of α' from Eq. (16):

$$f(I_{D,optimum}) = \frac{y}{1+y} \cdot f(I_C) \quad (27)$$

If the value of minimum power is P_{min} then from Eq. (8):

$$P_D - P_{min} = \tau \cdot (I_D - I_C) \cdot A_p \quad (28)$$

Substituting the value of b from Eq. (11) in the relation of τ given in Eq. (9):

$$\tau = \frac{(1+y) \cdot P_D \cdot \alpha}{m_D} \quad (29)$$

From Eqs. (28) and (29), the value of I_C can be expressed as:

$$I_C = I_D - \frac{P_D - P_{min}}{\tau \cdot A_p} = I_D - \frac{m_D \cdot (P_D - P_{min})}{(1+y) \cdot P_D \cdot \alpha \cdot A_p} \quad (30)$$

Substituting the value of m_D from Eq. (7):

$$\begin{aligned} I_C &= I_D - \frac{(\alpha \cdot I_D - \beta) \cdot (P_D - P_{min})}{(1+y) \cdot P_D \cdot \alpha} \\ &= I_D - \frac{(\alpha \cdot I_D - \beta) \cdot \left(1 - \frac{P_{min}}{P_D}\right)}{(1+y) \cdot \alpha} \end{aligned} \quad (31)$$

3.1. PTC based plant

The PTC field uses concentrated solar radiation incident on it to generate high temperature HTF, which is fed into heat exchanger as shown in Fig. 2. The heat exchanger will produce required steam to generate power. The collector

field heat gain (Q_{gain}) and efficiency (η) can be written as follows:

$$Q_{gain} = m_{oil} \cdot (h_2 - h_1) = \eta \cdot DNI \cdot \cos(\theta) \cdot A_p \quad (32)$$

$$\eta = \eta_o - U_L \cdot \left(\frac{T_m - T_a}{DNI \cdot \cos(\theta)} \right) \quad (33)$$

where m_{oil} is mass flow rate of the oil (kg/s), DNI is direct normal incidence radiation (W/m^2), θ is incidence angle, A_p is aperture area of the collector (m^2), $T_m = (T_1 + T_2)/2$, denotes mean temperature ($^{\circ}C$), T_a is ambient temperature ($^{\circ}C$), η_o is optical efficiency, and U_L is loss co-efficient based on aperture area ($W/m^2 K$). It may be noted that h_i and T_i denote the enthalpy and temperature at i -th state point, respectively.

Denoting the aperture effective DNI as I and the difference between T_m and T_a as ΔT :

$$Q_{gain} = m_{oil} \cdot (h_2 - h_1) = \eta \cdot I \cdot A_p \quad (34)$$

$$\eta = \eta_o - U_L \cdot \left(\frac{\Delta T}{I} \right) \text{ and } \eta_D = \eta_o - U_L \cdot \left(\frac{\Delta T}{I_D} \right) \quad (35)$$

where, η_D is the efficiency of collector at design condition. If the work input from Pump-I, Pump-II and the heat losses through piping are neglected then:

$$Q_{gain} = m_{oil} \cdot (h_2 - h_1) = m \cdot (h_5 - h_7) \quad (36)$$

where m is the mass flow rate of steam (kg/s). From Eqs. (34) and (36):

$$m \cdot (h_5 - h_7) = \eta \cdot I \cdot A_p \quad (37)$$

Substituting the value of collector efficiency from Eq. (35) in Eq. (37) and denoting the difference between h_5 and h_7 as Δh :

$$m \cdot \Delta h = \left(\eta_o - \frac{U_L \cdot \Delta T}{I} \right) \cdot I \cdot A_p \quad (38)$$

$$m = \frac{(\eta_o \cdot I - U_L \cdot \Delta T) \cdot A_p}{\Delta h} = (\alpha \cdot I - \beta) \cdot A_p \quad (39)$$

and the steam flow rate at design point (m_D) can be calculated as:

$$m_D = \frac{(\eta_o \cdot I_D - U_L \cdot \Delta T) \cdot A_p}{\Delta h} = (\alpha \cdot I_D - \beta) \cdot A_p \quad (40)$$

$$\text{where } \alpha = \frac{\eta_o}{\Delta h} \text{ and } \beta = \frac{U_L \cdot \Delta T}{\Delta h} \quad (41)$$

The temperature-enthalpy (T - h) diagram of heat exchanger is shown in Fig. 4. The duty of heat exchanger (Q_{hx}) is given as:

$$Q_{hx} = m_{oil} \cdot C_P \cdot (T_2 - T_3) = m_D \cdot (h_5 - h_4) \quad (42)$$

$$Q_{hx} = m_{oil} \cdot C_P \cdot (T_2 - T_1) = m_D \cdot (h_5 - h_7) \quad (43)$$

For a given value of temperature driving force (ΔT_{min}):

$$m_D \cdot (h_5 - h_w) = m_{oil} \cdot C_P \cdot \{T_2 - (T_P + \Delta T_{min})\} \quad (44)$$

where h_w is saturated water enthalpy at temperature T_P . Denoting the difference of h_5 and h_w as Δh_2 :

$$m_D = \frac{m_{oil} \cdot C_P \cdot (T_2 - T_P - \Delta T_{min})}{\Delta h_2} \quad (45)$$

Substituting the value of m_D from Eq. (45) into Eq. (43):

$$T_2 - T_1 = \frac{(T_2 - T_P - \Delta T_{min}) \cdot \Delta h}{\Delta h_2} \quad (46)$$

$$T_m = \frac{T_1 + T_2}{2} = T_2 - \left[\left(\frac{1}{2} \right) \cdot \left(\frac{\Delta h}{\Delta h_2} \right) \cdot (T_2 - T_P - \Delta T_{min}) \right] \quad (47)$$

$$\Delta T = T_2 - \left[\left(\frac{1}{2} \right) \cdot \left(\frac{\Delta h}{\Delta h_2} \right) \cdot (T_2 - T_P - \Delta T_{min}) \right] - T_a \quad (48)$$

3.2. LFR based plant

In case of LFR based plant similar kind of relation can also be obtained. The schematic of a typical LFR based CSP plant (without storage and auxiliary boiler) is shown in Fig. 5. Typically, at the outlet of LFR field a two-phase mixture is obtained. The mixture enters a drum, where saturated steam is directed to generate power. The collector field heat gain (Q_{gain}) and efficiency (η) can be written as follows:

$$Q_{gain} = m_{CL} \cdot (h_2 - h_1) = \eta \cdot I \cdot A_p \quad (49)$$

$$\eta = \eta_o - U_L \cdot \left(\frac{\Delta T}{I} \right) \text{ and } \eta_D = \eta_o - U_L \cdot \left(\frac{\Delta T}{I_D} \right) \quad (50)$$

where m_{CL} is the mass flow rate of steam passing through collector (kg/s). If the work input from Pump-I, Pump-II and Pump-III as well as heat losses through piping are neglected then:

$$Q_{gain} = m_{CL} \cdot (h_2 - h_1) = m \cdot (h_5 - h_7) \quad (51)$$

From Eqs. (49) and (51):

$$m \cdot (h_5 - h_7) = \eta \cdot I \cdot A_p \quad (52)$$

Substituting the value of collector efficiency in Eq. (52) and denoting the difference between h_5 and h_7 as Δh :

$$m = \frac{(\eta_o \cdot I - U_L \cdot \Delta T) \cdot A_p}{\Delta h} \quad (53)$$

The steam flow rate at design point (m_D) can be calculated as:

$$m_D = \frac{(\eta_o \cdot I_D - U_L \cdot \Delta T) \cdot A_p}{\Delta h} \quad (54)$$

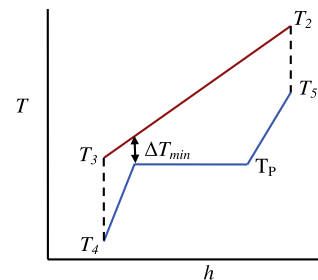


Fig. 4. Temperature-enthalpy (T - h) diagram for heat exchanger.

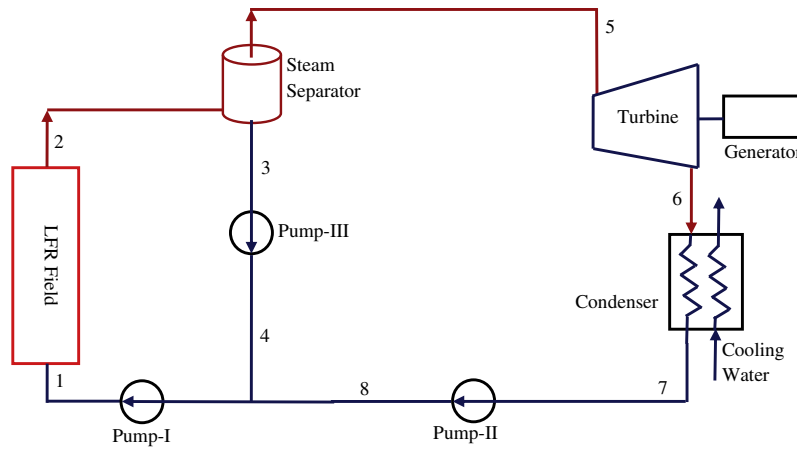


Fig. 5. Simplified process flow diagram of a LFR based solar thermal power plant.

where $\alpha = \frac{\eta_o}{\Delta h}$ and $\beta = \frac{U_L \cdot \Delta T}{\Delta h}$ (55)

It may be noted that the LFR inlet mass flow rate (m_{CL}) will generally remain fixed during the operation and as a result the LFR outlet dryness fraction as well as mean temperature difference (ΔT) will vary at part load.

The value of ΔT can be calculated by neglecting the pump work and assuming design point outlet dryness fraction of the LFR field (x_2).

$$h_1 = (1 - x_2) \cdot h_3 + x_2 \cdot h_7 \quad (56)$$

The value of LFR field inlet temperature (T_1) can be calculated from inlet pressure (P_1) and inlet enthalpy (h_1).

3.3. Thermodynamically optimum design radiation algorithm

Based on the results obtained in the previous sub-sections, an algorithm to calculate the optimum design radiation is proposed in this sub-section. The steps of proposed algorithm are as follows:

- *Step 1:* Specify the following data required for calculations: Ambient temperature (T_a), Optical efficiency of collector field (η_o), Loss co-efficient of collector field based on aperture area (U_L), Collector field outlet temperature (T_2) or outlet dryness fraction (x_2), Temperature driving force for heat exchanger (ΔT_{min}), Turbine inlet temperature and pressure (T_5 and P_5) and Internal losses of turbine (y).
- *Step 2:* Calculate the value of ΔT .
- *Step 3:* The optimum value of I_D is calculated through iterations as the value of I_C is dependent on I_D . Calculate the value of I_C using $I_D = I_{max}$ from equation (31) and get the value of $f(I_C)$ from duration curve of aperture effective DNI.
- *Step 4:* Calculate the value of $f(I_{D,optimum})$ from equation (27) and get the value of $I_{D,optimum}$ from duration curve of aperture effective DNI.

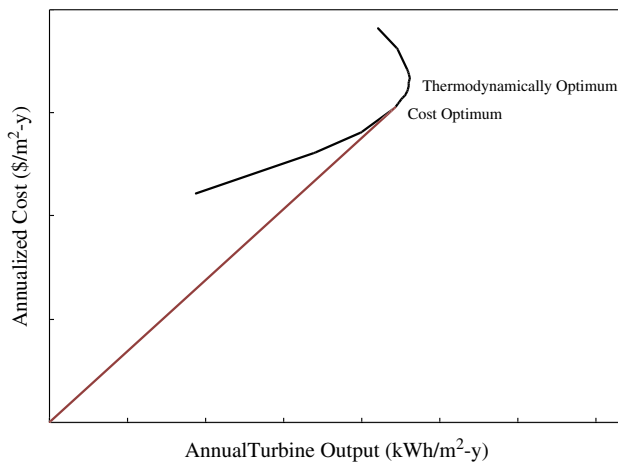


Fig. 6. Typical curve for annualized cost vs. annual turbine output per unit aperture area of collector.

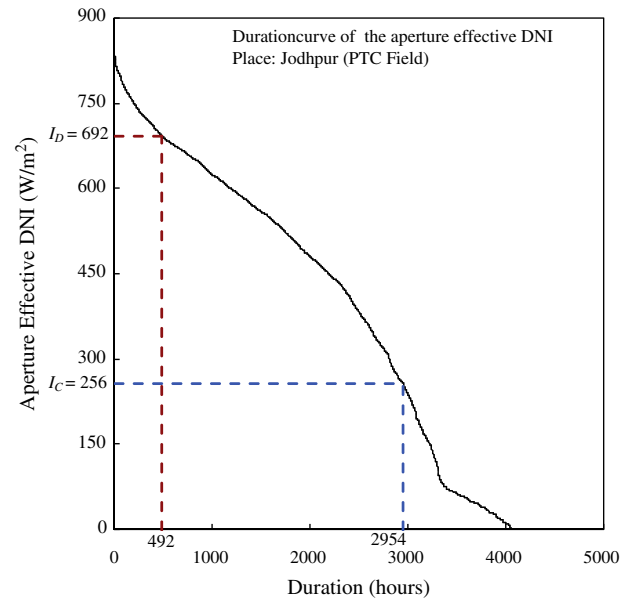


Fig. 7. Duration curve of aperture effective DNI for PTC field (Place: Jodhpur).

Table 3
Results for Example 1.

Parameter	Iteration 1	Iteration 2	Iteration 3	Optimum from simulation
I_D (W/m ²)	874	698	692	680
y	0.2	0.2	0.2	
$y/(1+y)$	0.1667	0.1667	0.1667	
I_C (W/m ²)	316.7	258	256	
$f(I_C)$	2759	2947	2954	
$f(I_{D,optimum})$	460	491	492	
$I_{D,optimum}$ (W/m ²)	698	692	692	

Table 4
Cost data for economic analysis.

<i>Investment</i>	
PTC field cost (US\$/m ²)	250
Heat exchanger cost (US\$/kW _{th})	41.67
Power block cost (US\$/kW _e)	980
Land cost (US\$/m ²)	1.67
Civil works cost (US\$)	$a \cdot kW_e + b \cdot (kW_e)^2$ $a = 169.3; b = -5.3 \times 10^{-4}$
Miscellaneous cost (US\$/kW _e)	183.33
<i>Operation and maintenance</i>	
Annual solar field component replacement cost	2.5% of solar field cost
Annual operating and maintenance cost	3% of equipment cost
<i>Financial parameters</i>	
Discount rate (%)	10
Lifetime (years)	30

Table 5
Data for Example 2.

Input parameter	Value/type
Collector field	Linear Fresnel Reflector (LFR)
Collector field cost (US\$/m ²)	166.67
Collector efficiency model parameters	$\eta_o = 0.6, U_L = 0.2$ (W/m ² -K)
Collector receiver height	13 m
No. of reflectors in each loop	8
Distance between two reflectors	2.378 m
Collector inlet pressure	45 bar
Collector outlet dryness fraction	0.5
Willans' line equation	Turbine power output: $P = a + b \cdot m$ where $a = -0.2 \cdot P_D$ (Mavromatis and Kokossis, 1998) $b = 686$ kJ/kg
Turbine inlet pressure	40 bar
Turbine inlet temperature	250.4 °C ($=T_{sat@40 \text{ bar}}$)

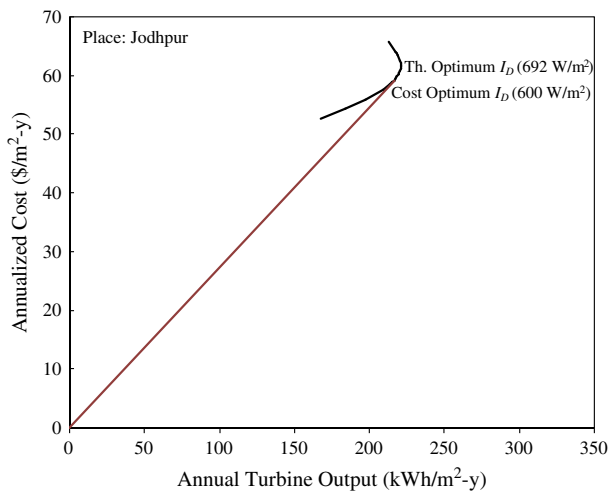


Fig. 8. Annualized cost vs. annual turbine output per unit aperture area of collector for PTC based plant.

- *Step 5:* Repeat *Step 3* using a value of I_D as $I_{D,optimum}$, calculated in *Step 4*. Repeat *Step 4* using the new value of I_C . The procedure is repeated till guess value of I_D will be equal to calculated value of $I_{D,optimum}$.

3.4. Cost optimum design DNI

Higher value of design DNI leads to higher design mass flow rates and higher rated power output (see Tables 2 and

6), which leads to increase in the cost of power block and auxiliaries. Therefore, the value of cost optimum design radiation is always lower than the thermodynamically optimum value. As discussed in previous sub sections, the value of design mass flow rates and rated power output, per unit aperture area of the collector, can be calculated from Eqs. (7) and (15), respectively. Subsequently, the annualized cost, per unit area of the collector, can also be calculated. Typical plot of annualized cost vs. annual turbine output per unit aperture area of the collector is given in Fig. 6. A tangent line to the plot that passes through the origin represents the cost optimal design DNI, as the slope of the tangent line represents the cost of energy. It may be noted that the design DNI, corresponding to the maximum annual turbine output, is the thermodynamically optimum. It can further be observed from Fig. 6 that the thermodynamically optimum design DNI is always higher than the cost optimal design DNI.

4. Illustrative examples

Applicability of the proposed methodology is demonstrated through illustrative examples.

4.1. Example 1: PTC based plant

The proposed methodology is applied to the data, given in Table 1. Duration curve of aperture effective DNI for

Table 6

Simulation results for a LFR based solar thermal power plant without hybridization and thermal storage.

Design DNI (W/m ²)	Design mass flow rate through collector (kg/s)	Design mass flow rate through turbine (kg/s)	Design turbine output (MW)	Annual turbine output (MW h/y)	Net annual output (MW h/y)
540	2.22	1.110	0.635	1477.5	1453.3
550	2.27	1.135	0.649	1487.5	1463.0
570	2.36	1.180	0.676	1504.7	1479.5
580	2.41	1.205	0.689	1509.9	1484.8
600	2.50	1.250	0.715	1520.5	1494.8
610	2.55	1.275	0.729	1522.1	1496.3
620	2.60	1.300	0.743	1523.3	1497.2
630	2.64	1.320	0.755	1524.0	1497.9
640	2.69	1.345	0.769	1525.1	1498.8
650	2.73	1.365	0.780	1522.7	1496.4
660	2.78	1.390	0.795	1521.0	1494.2
680	2.87	1.435	0.821	1513.0	1486.0
690	2.92	1.460	0.834	1507.9	1480.9
750	3.19	1.595	0.912	1463.2	1436.0

Jodhpur, India is shown in Fig. 7. Cosine of the incidence angle, for PTC rotated about a horizontal north-south axis with continuous east-west tracking, can be calculated from Sukhatme (2002). The results based on proposed methodology are shown in Table 3. Optimum value of I_D is

692 W/m² which is much closer to the value obtained through detailed multiple simulations (680 W/m²). As mentioned earlier the nature of the power output curve is not very sharp near the maximum and the first iteration results are also sufficient.

Effect of design radiation on cost of energy is studied using the cost data given in Table 4 (Montes et al., 2009; IIT Bombay, 2012; Krishnamurthy et al., 2012). Fig. 8

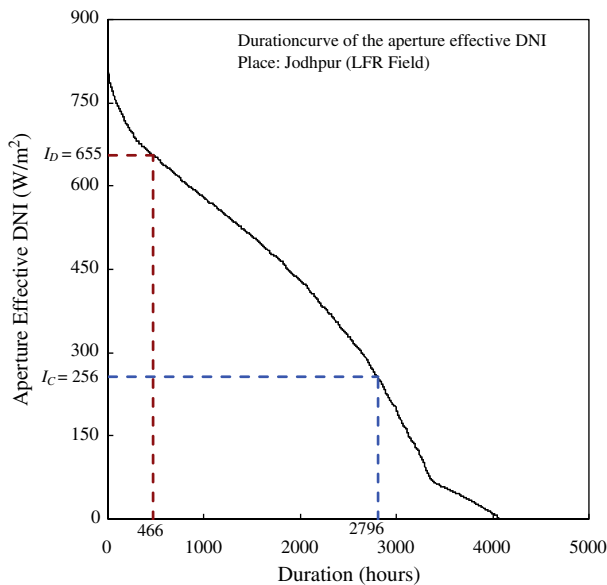


Fig. 9. Duration curve of aperture effective DNI for LFR field (Place: Jodhpur).

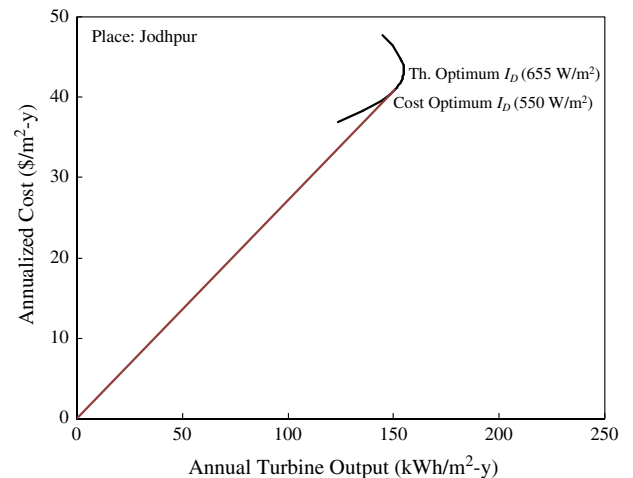


Fig. 10. Annualized cost vs. annual turbine output per unit aperture area of collector for LFR based plant.

Table 7

Results for Example 2.

Parameter	Iteration 1	Iteration 2	Iteration 3	Optimum from simulation
I_D (W/m ²)	806	660	655	640
y	0.2	0.2	0.2	
$y/(1+y)$	0.1667	0.1667	0.1667	
I_C (W/m ²)	306.6	258	256.3	
$f(I_C)$	2603	2787	2796	
$f(I_{D,optimum})$	434	465	466	
$I_{D,optimum}$ (W/m ²)	660	655	655	

shows the plot of annualized cost vs. annual turbine output per unit aperture area of collector for PTC based plants. The value of cost optimum I_D for PTC based plant is 600 W/m^2 which is lesser than the thermodynamically optimum value (692 W/m^2), as expected. The corresponding value of minimum cost of energy is $0.273 \text{ US\$/kW h}$. Approximate method based on frequency distribution of aperture effective DNI proposed by Quaschnig et al. (2002), for finding cost optimum design radiation of PTC based CSP plants, is also applied for metrological data of Jodhpur. The range for cost optimum design radiation based on this method is $550\text{--}590 \text{ W/m}^2$, which is slightly lower than the value (600 W/m^2) obtained through proposed methodology.

4.2. Example 2: LFR based plant

The additional data required for this example are given in Table 5. Simulations are performed in the solar thermal simulator (IIT Bombay, 2012) and annual turbine and net power outputs, at different design DNI, are given in Table 6. The maximum annual turbine output of 1525.1 MW h and the maximum net annual output of 1498.8 MW h are obtained with 640 W/m^2 of design DNI. Design DNI within the range $580\text{--}680 \text{ W/m}^2$, the

net annual output of the plant remains within 1% of the maximum.

Duration curve of aperture effective DNI for Jodhpur, India is shown in Fig. 9. In case of LFR, the incidence angle (θ) is taken as mean incidence angle (θ_{mean}), which is the average incidence angle of each reflector rows. The method to calculate $\cos(\theta_{mean})$ for LFR rotated about a horizontal north-south axis with continuous east-west tracking is given by Nixon and Davies (2012). The proposed methodology is applied to the given data and results are shown in Table 7. It may be noted that the optimum value of I_D is 655 W/m^2 , very close to the value obtained through multiple simulations (640 W/m^2).

Effect of design radiation on cost of energy is studied using the cost data. Fig. 10 shows the plot of annualized cost vs. annual turbine output per unit aperture area of collector for LFR based plant. The value of cost optimum I_D is 540 W/m^2 which is lesser than the thermodynamically optimum value (655 W/m^2), as expected. The corresponding value of minimum cost of energy is $0.272 \text{ US\$/kW h}$.

5. Parametric study using proposed methodology

Effect of various important parameters on optimum value of aperture effective DNI (I_D) is discussed in this

Table 8
Effect of the turbine parameters on optimum aperture effective DNI.

Parameter	PTC based plant					LFR based plant			
	Th. optimum I_D (W/m^2)	Annual turbine output ($\text{kW h/m}^2 \text{ y}$)	Cost optimum I_D (W/m^2)	Cost of energy ($\text{US\$/kW h}$)		Th. optimum I_D (W/m^2)	Annual turbine output ($\text{kW h/m}^2 \text{ y}$)	Cost optimum I_D (W/m^2)	Cost of energy ($\text{US\$/kW h}$)
Internal losses of turbine (γ)	0.1	731	231.1	620	0.265	689	162.1	580	0.263
	0.2	692	221.2	600	0.273	655	154.6	550	0.272
	0.3	670	213.1	585	0.28	631	148.5	530	0.278
Isentropic efficiency of turbine at design	0.65	692	221.2	600	0.273	655	154.6	550	0.272
	0.80	692	272.3	590	0.235	655	190.3	540	0.235

Table 9
Thermodynamically and cost optimum value of aperture effective DNI for ten different places of India.

Place	Annual DNI ($\text{kW h/m}^2 \text{ y}$)	PTC based plant				LFR based plant			
		Th. optimum I_D (W/m^2)	Annual turbine output ($\text{kW h/m}^2 \text{ y}$)	Cost optimum I_D (W/m^2)	Cost of energy ($\text{US\$/kW h}$)	Th. optimum I_D (W/m^2)	Annual turbine output ($\text{kW h/m}^2 \text{ y}$)	Cost optimum I_D (W/m^2)	Cost of energy ($\text{US\$/kW h}$)
Jodhpur	1942	692	221.2	600	0.273	655	154.6	550	0.272
Hyderabad	1869	771	215.8	660	0.29	737	152.0	610	0.288
Jaipur	1759	669	196.2	570	0.305	639	137.2	530	0.303
Bangalore	1569	770	172.2	620	0.355	728	120.5	560	0.353
Ahmedabad	1635	678	180.6	570	0.33	643	126.1	530	0.33
New Delhi	1538	618	163.1	520	0.352	587	112.6	470	0.352
Nagpur	1532	667	171.5	570	0.347	638	119.5	510	0.345
Chennai	1418	692	159.3	560	0.373	655	111.8	530	0.37
Mumbai	1346	636	144.6	500	0.395	603	100.6	460	0.393
Kolkata	1095	509	111.7	430	0.487	483	76.6	400	0.488

section and the data given in previous subsections are used for parametric study. The results of variation in turbine parameters on optimum value of I_D are shown in Table 8. It shows that, the value of $f(I_{D,optimum})$ increases with increase in value of internal losses of turbine and subsequently the value of optimum I_D decreases. There is no significant change in the value of optimum I_D with variation in the isentropic efficiency of turbine at design ($\eta_{is,D}$). However, the corresponding values of annual turbine output increases and cost of energy decreases with increase in isentropic efficiency of turbine at design, as expected. It was also observed that, there is no significant change in the value of optimum I_D with variation in optical efficiency of collector field, overall heat loss coefficient of collector field, turndown ratio of turbine and inlet pressure as well as temperature of turbine.

The value of optimum I_D strongly depends on its duration curve. Table 9 presents the value of thermodynamically and cost optimum I_D for ten different places of India. It shows that, the value of optimum I_D for LFR based plant will be always lesser than the PTC based plant. This is mainly due to the higher cosine effect in case of LFR field compared to the PTC field and that can be observed from the duration curves of I_D , as shown in Fig. 11. It may be noted that the value of annual DNI is almost same for the place New Delhi and Nagpur but the difference in thermodynamically optimum value of I_D is significant. The same can be observed between Ahmedabad and Bangalore as well as Hyderabad and Jodhpur. This is mainly due to the shape of duration curve of I_D , as shown in Fig. 12. The cost optimum value of I_D as well as cost of energy is also strongly dependant on the duration curve of I_D . Depending on the data used for analysis, the place Jodhpur is having least cost of energy.

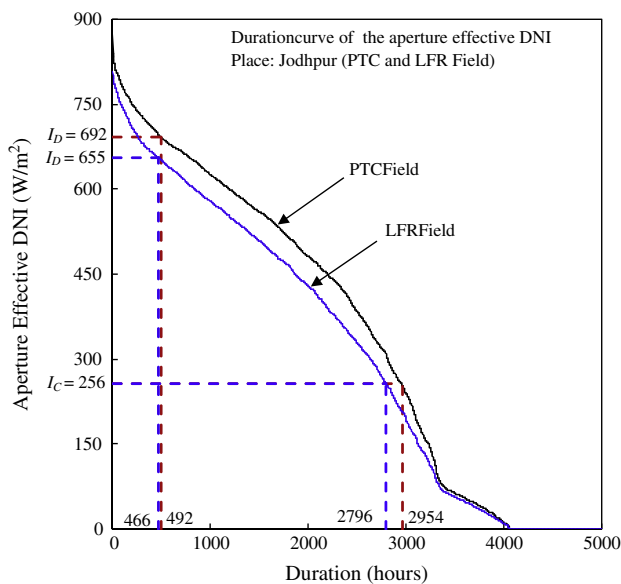


Fig. 11. Duration curve of aperture effective DNI for PTC and LFR field (Place: Jodhpur).

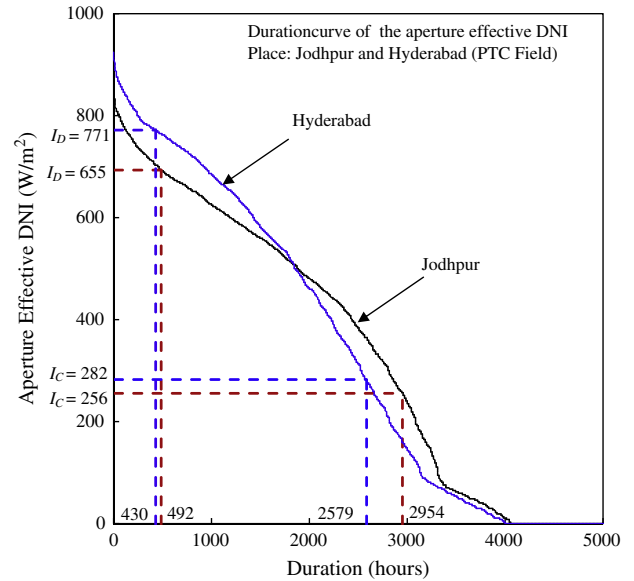


Fig. 12. Duration curve of aperture effective DNI for Jodhpur and Hyderabad (Collector field: PTC).

6. Case study

Under the initiative of IIT Bombay, a one megawatt CSP plant was proposed in the year 2009 and currently being commissioned in Gurgaon (near New Delhi), India that integrates a LFR field for DSG and PTC field for HTF (Desai et al., 2013). The plant is developed as a part of the project titled ‘Development of a Megawatt-scale Solar Thermal Power, Testing, Simulation and Research Facility’, sponsored by the Ministry of New and Renewable Energy, Government of India. The simplified PFD of the plant is shown in Fig. 13 (Desai et al., 2013). The plant integrates a LFR field for DSG and PTC field for HTF (Therminol VP1). At the outlet of LFR field a two-phase mixture is obtained and the mixture enters into a steam separator, where the saturated steam (44 bar and 256.1 °C) is directed for superheating and the liquid is re-circulated to the LFR field inlet. The PTC field generates high temperature oil at 390 °C, which is fed into the heat exchanger for both steam generation and superheating the combine steam.

The plant has a gross capacity of 1 MWe at DNI of 600 W/m². As the cost data at initial design time of the plant were not available and storage is very small in amount (for about 20 min) as well as the main purpose of setting up such a plant is research and development, not commercial, the plant is designed with thermodynamically optimum values of design radiation given by the proposed methodology. In case of New Delhi, the thermodynamically optimum values of design radiation for PTC based and LFR based plant are 618 W/m² and 587 W/m² (see Table 8), respectively. Therefore, the design radiation is taken as 600 W/m² (approximately the average of both values). At design condition, the PTC field supplies about 60% of the required heat, while the LFR field supplies

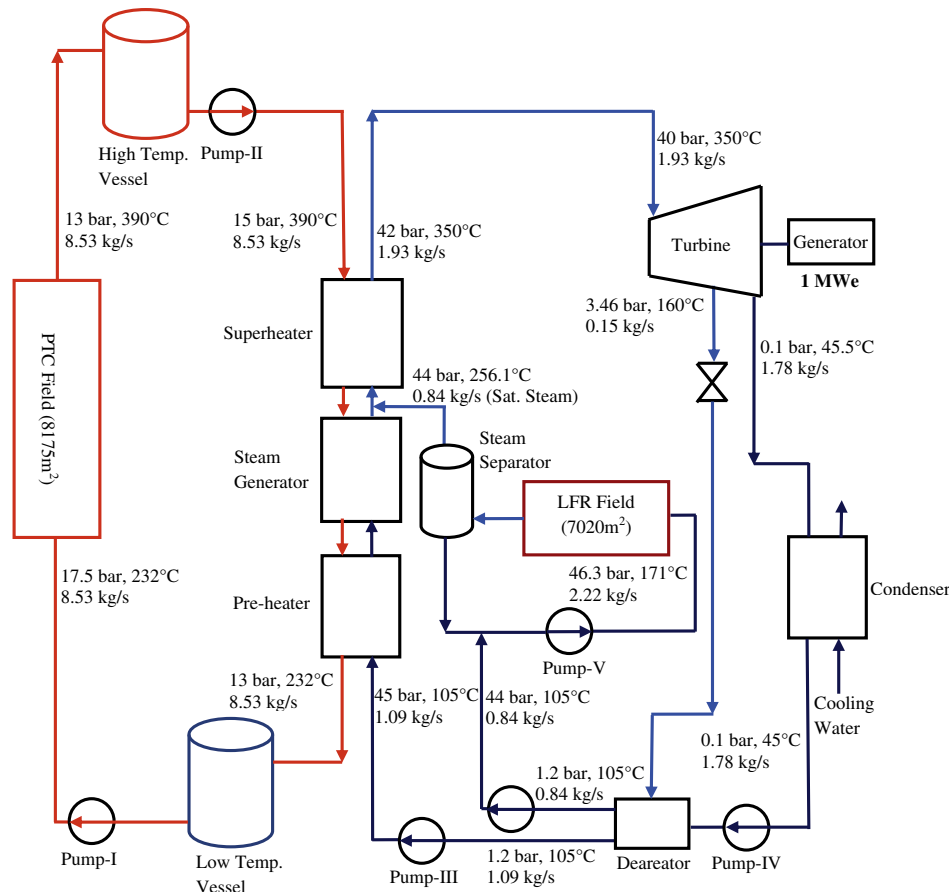


Fig. 13. Simplified process flow diagram of 1 MWe solar thermal power plant (Desai et al., 2013).

Table 10
Comparison of optimum and actual aperture area for various PTC based CSP plants.

Name of the plant	Gross output (MW)	Parameters					Th. optimum I_D (W/m^2)	Cost optimum I_D (W/m^2)	Cost optimum aperture area of collector field (in $10^5 m^2$)	Actual aperture area (in $10^5 m^2$) (NREL, 2014)	Remarks
		η_o	U_L ($W/m^2 K$)	P_5 (MPa)	T_5 ($^{\circ}C$)	$\eta_{is,D}$					
Godawari Project (India)	50	0.75	0.1	100	370	0.87	692	590	3.62	3.92	100% Solar plant
Solnova 1 (Spain)	50	0.75	0.1	100	370	0.87	790	650	3.26	3.00	Hybrid plant
SEGS VII (USA)	35	0.75	0.1	100	370	0.87	850	710	2.08	1.94	Hybrid plant

the balance about 40% of the required heat. It may be noted that, the plant is designed without any auxiliary boiler.

To show the applicability of the proposed methodology, different operational PTC based CSP plants from different parts of the world are considered. As design radiation for these plants are not reported in the literature, installed aperture area of these plants are compared against the aperture area calculated based on the cost optimal design radiation of these plants and these results are tabulated in Table 10. It may be observed that the actual aperture

area of collector field is much closer to the cost optimum aperture area calculated using proposed methodology. Almost all CSP plants in USA and Europe are hybrid plants with fossil fuel backup and therefore higher design radiation (lower aperture area of the collector field) compared to the design radiation obtained through proposed methodology is expected. On the other hand, most of the CSP plants coming up in India are without hybridization and therefore lower design radiation (higher aperture area of the collector field) compared to the design radiation obtained through proposed methodology is expected.

7. Conclusions

Conceptual design of a CSP plant includes type and size of solar field, power generating cycle and the working fluid, sizing of the power block, etc. One of the most important parameter for conceptual design of a CSP plant is to determine the design radiation of the plant. Due to daily and seasonal variation of radiation, a simple methodology is presented to determine the optimum design radiation, for which the annual power generation per unit aperture area of collector is the maximum, for CSP plant without hybridization and storage. The proposed methodology accounts for the characteristics of the collector field as well as turbine characteristics and does not require any simulations. A methodology to determine optimum design radiation that minimizes cost of electricity generation is also presented. There is no such analytical methodology reported in the literature. The presented methodology can be applied to CSP plants with any technology.

It is observed through illustrative examples that the value of optimum design radiation obtained through proposed methodology is very close to that obtained with detailed multiple simulations. The value of optimum design radiation is mainly dependent on the shape of the duration curve for aperture effective DNI, collector and turbine characteristic parameters. Comparison between the thermodynamically optimum values of design radiation with cost optimum values shows that the latter is lower than the former. It is also observed that the value of optimum design radiation for LFR based plant is always lower than the PTC based plant.

Parametric study shows that the higher internal losses of turbine will decrease the value of optimum design radiation. However, there is no significant change in the value of optimum design radiation with variation in optical efficiency of collector field, overall heat loss coefficient of collector field, turndown ratio of turbine, isentropic efficiency of turbine at design and inlet pressure as well as temperature of turbine. It may be noted that the present study includes CSP plants without storage and auxiliary boiler. Present research is directed towards optimizing design DNI for CSP plants with storage unit.

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